

Performance comparison of a novel configuration of beta-type Stirling engines with rhombic drive engine



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ARTICLE INFO

Article history:

Received 4 October 2013

Accepted 22 November 2013

Available online 16 December 2013

Keywords:

Stirling engine

Beta type

Rhombic drive

Lever driven

ABSTRACT

This study presents a beta type Stirling engine mechanism and its performance analysis. The displacer motion of the engine is performed by a lever mechanism. The performance of the engine was investigated via comparing with a rhombic-drive engine possessing an equal sided rhombic. Comparison was made for kinematic behaviors, power and thermal efficiency. For comparison; the piston swept volume, the inner heat transfer area, the hot and cold end temperatures, the inner heat transfer coefficient, charge pressure and dead volumes were kept equal for both engines. As working fluid the helium was used. Thermodynamic treatments of engines were performed via the nodal analysis. The power of the lever driven engine was found to be greater than the power of the rhombic drive engine. Under the equal charge pressure, the thermal efficiency of the lever driven engine was found to be lower than the efficiency of the rhombic drive engine however, under the equal working fluid mass the thermal efficiency of the lever driven engine was found to be greater than that of the rhombic drive engine. The external volume and mass of the lever driven engine is lower than the rhombic drive engine.

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1. Introduction

Stirling engines are externally heated engines capable of using all kinds of heats such as combustion heat, solar heat, and radioisotope heat. The other important feature of the Stirling engine is that it is a regenerative engine having a theoretical thermal efficiency of equal to the Carnot efficiency [1]. The engine has some more advantages such as less harmful exhaust emissions, and relatively lower noise characteristic [2,3]. Because of these properties, before 1980 the Stirling engine had attracted the attention of engineers. As well as academicians, the private companies had conducted some comprehensive investigation to develop Stirling type vehicle engines, however, the specific powers of the engines developed were too low to use at a vehicle. The magnitude of the power generated by the engines could be not controlled efficiently according to the power requirement of the vehicles. Because of these disadvantages the investigation aiming to develop Stirling type vehicle engines were terminated after 1980 [4].

Another application field of Stirling engines was the solar energy investigations which were initiated by Malik and Parker in 1962 [4]. In solar energy investigations mostly the free-piston type Stirling engines were used however, kinematic engines were also used [5–8]. The governmental solar energy plants set by DOE in United states provided efficiencies of reaching to 30% however the price of the energy generated by these plants was not competi-

itive with fossil energy [9]. Therefore the Stirling based solar energy investigations were terminated after 1990. Then, some academicians intended to develop some micro solar energy plant based on Low Temperature Difference (LTD) Stirling Engines [10–13]. The investigations intending to develop micropower plants based on Stirling engine are still undergoing [14,15]. The most recent application of the Stirling engine is the Micro Combined Heat Power systems (MCHP) [16–20]. MCHP is a domestic plant fueled by natural gas. Simultaneously the system generates heat and electricity where the heat is used at home where the plant is set but the electricity is sold to the national network.

The engines named as LTD are constructed in two different manner. In one of them the transition of the working fluid between the hot and cold volumes of the engine is accomplished through a regenerative channel between the displacer and its cylinder which can be named as internally connected. The channel acts as heater and cooler as well as regenerator but the regeneration may be too little. The shaft power of the most of this type engines did not exceed 700 W per liter swept volume [21,22].

In engines with externally connected expansion and compression volumes, the connection is made with a tube bundle. The heater cooler and regenerator are separate components but adjacent to each other. The efficiency of the regenerator is high enough. The shaft power may reach to 3 kW per liter swept volume despite the same heat consumption [23]. The power difference between these two types of engines is caused by the regenerative heat. To increase the power density of Stirling engines the working fluid pressure is increased. The volume inside the engine block is used

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Nomenclature

A_c	crosscut area of the cylinder (m^2)	u_c	length of cylinder (m)
A_i	nodal values of heat transfer surface (m^2)	u_d	length of displacer (m)
A_r	crosscut area of the displacer rod (m^2)	V_h	expansion volume (m^3)
c_p	specific heat at constant pressure (J/kg K)	V_c	compression volume (m^3)
c_v	specific heat at constant volume (J/kg K)	V_{cr}	compression volume of rhombic driven engine (m^3)
d	length of lever arm connected to intermediate arm (m)	V_{hr}	expansion volume of rhombic driven engine (m^3)
h	length of intermediate arm (m)	V_{ri}	nodal volumes of regenerator (m^3)
h_i	nodal values of convective heat transfer coefficient ($\text{W/m}^2 \text{K}$)	y_{pt}	distance between crank centre and piston top (m)
h_d	displacer length (m)	y_{db}	distance between crank centre and displacer bottom (m)
h_p	piston length (m)	y_{dt}	distance between crank centre and displacer top (m)
H_{in}	enthalpy flow to nodal volumes (J/kg)	y_{ct}	distance between crank centre and cylinder top (m)
H_{out}	enthalpy flow from nodal volume (J/kg)	z	horizontal distance between crank centre and lever centre (m)
l_d	length of displacer connecting rod (m)	β	angle made by lever arm with vertical (rad)
l_R	length of displacer rod (m)	β_r	see Fig. 2
ℓ_r	the side length of rhombic (m)	σ	angle made by lever arm with intermediate arm (rad)
m_t	total mass of working fluid (kg)	ϑ	angle made by displacer connecting rod with vertical (rad)
m_i	nodal values of working fluid mass (kg)	ψ	angle made by piston rod with vertical (rad)
p	working volume pressure (Pa)	θ	crankshaft rotation (rad)
r	radius of crankshaft (m)	θ_r	see Fig. 2
R_r	radius of crankshaft of rhombic driven engine (m)	ϕ	angle between lever arms (rad)
R_g	gas constant (J/kg K)	Δm_i	variation of nodal mass within a time step (kg)
T_c	cold end temperature (K)	ΔT_i	variation of nodal temperature within a time step (K)
T_h	hot end temperature (K)	ΔQ_R	heat exchange between the regenerator and working gas in a time step
T_{ri}	nodal temperatures of regenerator (K)	Δt	period of time steps (s)
T_i	nodal values of working fluid temperature (K)	Ω	a dummy constant
u	vertical distance between crank centre and lever centre (m)	χ	length of lever arm connected to displacer connecting rod (m)
u_{pt}	distance between crank centre and piston top (m)	λ	length of piston rod (m)
u_{db}	distance between crank centre and displacer bottom (m)		
u_{dt}	distance between crank centre and displacer top (m)		
u_{dr}	length of displacer rod (m)		

as working fluid reservoir. In Stirling engines escaping of working fluid from the block is a serious problem. To solve this problem totally hermetic engine configurations are thought where the power is converted to electricity inside the engine block [24–27]. In this case however, the startup mechanism becomes complicated. Chen et al. [26] suggested a magnetic coupling mechanism to extract the motion outside the engine block. A comprehensive review of studies involving technological developments made before 2008 is given by Thombare and Verma [28].

Despite that the solar energy investigations conducted in last decade has been focused on solar cells, the academic studies conducted on Stirling based solar energy investigations are continuing as well. Aksoy and Karabulut [14] established a microsolar energy conversion system consisting of a Stirling engine and a Fresnel lens. The engine used is beta type and its expansion and compression volumes are internally connected. The solar radiation was focused into a cavity taking part above the displacer cylinder. In a test conducted with a copper cavity a 3% total conversion efficiency from solar radiation to mechanical work was obtained. In another solar energy test conducted with an aluminum cavity, the workers obtained a total conversion efficiency of 5.6%. Efficiencies reaching to 10% were also observed but the test duration was less than 15 min. Compared to the system previously established by the authors [8], this system was found to be more promising. For the current situation for solar cells some conversion efficiencies exceeding 24% were introduced [29]. This progress can reduce the interest to Stirling based solar energy conversion systems however; the interest to Stirling engines should not decline because of that it has too much application fields in particular the micro-CHP systems [16,19].

In flame or hot gas heated Stirling engines, the heat transfer from hot gas to heater of the engine is one of the most important reasons of low efficiency. In piston-displacer type engines, the hot end is located generally above the displacer cylinder and a forced circulation equipment is required for the heat transfer between the hot gas and the hot end [8,14]. Forced circulation equipment which increase cost, mass and volume of the engine, is a complex system for Stirling engines. For better transfer of the heat from a flame to the hot end of a Stirling engine the hot end of the displacer cylinder of the engine should be directed downward. The engines containing oil in its crankcase are not able to work at down turned position. It is seen essential to develop of an engine running without lubrication for reversing the engine. Karabulut et al. [3,21] developed a lever mechanism with sliding bearing. In this mechanism, one of the lever arms linked with crank by means of a sliding bearing. The lubrication necessity of this bearing prevents reversing the engine.

In this study a novel configuration of beta-type kinematic Stirling engine requiring no lubricant in its crankcase, structurally simple, easy in manufacturing, competitive with rhombic drive Stirling engines in power density and thermal efficiency, has been presented and analyzed thermodynamically.

2. Mechanism and working procedure of the engine

Fig. 1 illustrates the mechanical arrangement of the engine and some of symbols used in the paper. The displacer motion of the engine is accomplished by a lever mechanism. The mechanism of the

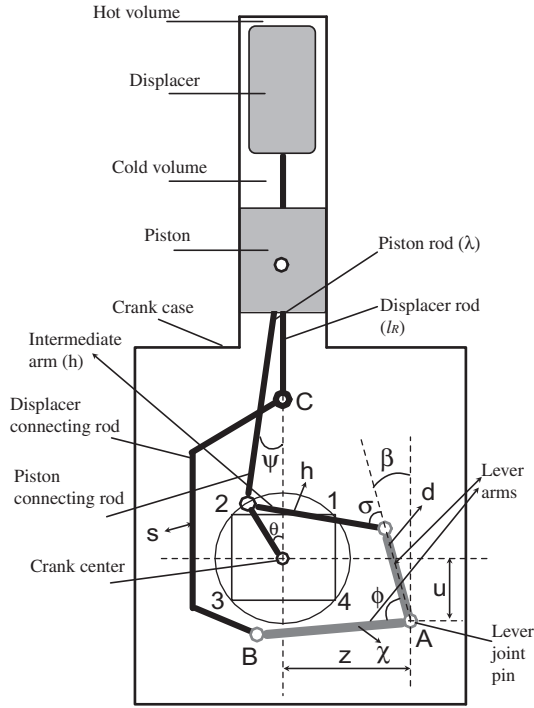


Fig. 1. Schematic illustration of the lever type Stirling engine.

engine consists of a cylinder, a crank case, a displacer, a displacer rod, a displacer connecting rod, a piston, a piston connecting rod, a crankshaft, a lever with two arms and an intermediate arm. The lever is jointed to the casing of the engine. One arm of the lever is connected to the crank pin via an intermediate arm. The other arm is connected to the displacer connecting rod which is a curved element. The angle between two arms of the lever was optimized as 87.5°. While the crankshaft turns around the crank center, it drives the lever arm forth and back around the joint pin via intermediate arm. The other arm of the lever operates the displacer via the displacer connecting rod.

With respect to Fig. 1, the running direction of the engine is anticlockwise. While the crank pin moves from the position 1 to the position 2, the piston is kept about at the top dead center of its stroke. The displacer moves from up to down and causes the working fluid to move from the compression volume to the expansion volume so as the heating process is performed. During this period since the displacement of the piston is little, the process may be assumed as a constant volume heating process. While the crank pin moves from the position 2 to the position 3, the piston moves from up to down and generates work. During the half of this process the displacer keeps going down but, about at the mid of the period, the down motion of the displacer terminates. After this instance, the working fluid starts to move from the expansion volume to the compression volume. While the crank pin moves from the position 3 to the position 4, the piston remains about the bottom dead center of its stroke. The displacer moves from down to up and causes the working fluid to move from the expansion volume to the compression volume. This process is a constant volume cooling process. While the crank pin moves from the position 4 to the position 1, the piston moves up and compresses the working fluid. During the half of this period the displacer keeps going up, about at the mid of the period it turns back. When the crank pin arrived to the position 1, the cycle is completed and the most of the working fluid is compressed in the compression volume.

3. Mathematical modeling

3.1. Kinematic relations of the engine with lever driven displacer

Fig. 1 illustrates the symbols used in the following equations. The crank angle is an independent variable of the analysis. According to Fig. 1,

$$\beta = \arcsin \left[\frac{z - h \sin(\sigma + \beta) + r \sin \theta}{d} \right] \quad (1)$$

$$\sigma = \arccos \left[\frac{u - d \cos \beta + r \cos \theta}{h} \right] - \beta \quad (2)$$

$$\vartheta = \arcsin \left(\frac{\chi \sin(\beta + \phi) - z}{\ell_d} \right) \quad (3)$$

$$\psi = \arcsin \left(\frac{r}{\lambda} \sin \theta \right) \quad (4)$$

Assuming that the crank center is reference point, the place of the piston and displacer may be described as

$$y_{pt} = r \cos \theta + \lambda \cos \phi + h_p \quad (5)$$

$$y_{db} = -u + \chi \cos(\beta + \phi) + l_d \cos \vartheta + l_r \quad (6)$$

$$y_{dt} = -u + \chi \cos(\beta + \phi) + l_d \cos \vartheta + l_r + h_d \quad (7)$$

The instantaneous values of the hot and cold volumes of the lever driven engine may be calculated as,

$$V_h = (y_{ct} - y_{dt})A_c \quad (8)$$

$$V_c = (y_d - y_p)(A_c - A_r) \quad (9)$$

3.2. Kinematic relations of the rhombic drive engine

The kinematic relations below are derived for an engine with equal sided rhombic which is 6.66 cm. This rhombic is preferred because of that it provides a higher work generation and better performance. According to Fig. 2,

$$\beta_r = \arcsin \left(\frac{1}{2} - \frac{R_r}{\ell_r} \sin \theta_r \right) \quad (10)$$

Assuming that the center of gears is the reference point, the place of the displacer and the piston may be described as,

$$u_{pt} = R_r \cos \theta_r + \ell_r \cos \beta_r + u_{pr} + \frac{h_p}{2} \quad (11)$$

$$u_{db} = R_r \cos \theta_r - \ell_r \cos \beta_r + u_{dr} \quad (12)$$

$$u_{dt} = R_r \cos \theta_r - \ell_r \cos \beta_r + u_{dr} + u_d \quad (13)$$

The hot and cold volume of the rhombic driven engine may be calculated as,

$$V_{cr} = (A_c - A_r)(u_{db} - u_{pt}) \quad (14)$$

$$V_{hr} = A_c(u_c - u_{dt}) \quad (15)$$

3.3. Thermodynamic relations

Working fluid pressure in all of the nodal volumes is assumed as equal. The instantaneous values of the working fluid pressure are calculated by [14,26],

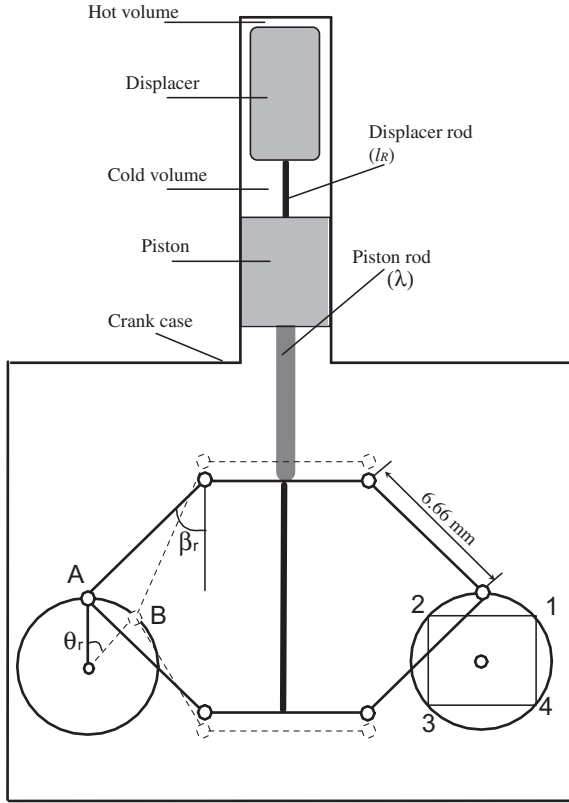


Fig. 2. Schematic illustration of the Rhombic driven Stirling engine.

$$p = \frac{m_i R_g}{\frac{V_c}{T_c} + \frac{V_h}{T_h} + \sum_{i=3}^{n-2} \frac{V_{Ri}}{T_{Ri}}} \quad (16)$$

For the calculation of temperature variation during a time step in nodal volumes, the first law of thermodynamics given for open systems

$$\Delta T_i (m_i c_v + \Omega) = h_i A_i (T_{w,i} - T_i) \Delta t - \Delta m_i c_v T_i + (H_{in} - H_{out})_i - p \Delta V_i + \Delta T_i \Omega \quad (17)$$

is used. In the last equation $(H_{in} - H_{out})_i$ is the enthalpy flow in and out of the nodal volumes and, it may be calculated by [14,26,27],

$$(H_{in} - H_{out})_i = -c_p \frac{T_{i+1} + T_i}{2} (\Delta m_{i+1} + \Delta m_{i+2} + \dots + \Delta m_n) - c_p \frac{T_{i-1} + T_i}{2} (\Delta m_1 + \Delta m_2 + \dots + \Delta m_{i-1}) \quad (18)$$

From Eq. (17) the temperature variation in a nodal volume is chosen as,

$$\Delta T_i = [h_i A_i (T_{w,i} - T_i) \Delta t - \Delta m_i c_v T_i + (H_{in} - H_{out})_i - p \Delta V_i + \Delta T_i \Omega] / (m_i c_v + \Omega) \quad (19)$$

In last equation Ω is an optional constant. If Ω is chosen as zero, numerical solution of the equations does not converge to stable values. Preferably Ω is chosen as 1. Instantaneous values of mass, m_i , are calculated by using the general state equation of perfect gases.

For a time step, the heat exchange between the regenerator and working gas is calculated by,

$$\begin{aligned} \Delta Q_R = & hA_1 \Delta t (T_{w1} - T_1) + hA_2 \Delta t (T_{w2} - T_2) + hA_3 \Delta t (T_{w3} \\ & - T_3) + hA_4 \Delta t (T_{w4} - T_4) + hA_5 \Delta t (T_{w5} - T_5) \\ & + hA_6 \Delta t (T_{w6} - T_6) + hA_7 \Delta t (T_{w7} - T_7) + hA_8 \Delta t (T_{w8} \\ & - T_8) + hA_9 \Delta t (T_{w9} - T_9) + hA_{10} \Delta t (T_{w10} - T_{10}) \end{aligned} \quad (20)$$

where the regenerator was divided into 10 nodal volumes as seen in Fig. 3.

4. Results and discussion

In both engines, the space occupied by working fluid was divided into 12 nodal volumes. The first of the nodal volume is the hot space consisting of expansion volume and heater volume as seen in Fig. 3. The last of nodal volumes is the cold space consisting of compression volume and cooler volume. The other nodal volumes take part in regenerator. Boundary temperature of nodal volumes varies as seen in Fig. 3 and constant in time. The specific values used in the analysis for both engines are seen in Table 1.

Fig. 4a and Fig. 4b illustrate the variations of compression, expansion and total volumes of lever-drive and rhombic-drive engines respectively. In both figures the period $45^\circ \leq \theta \leq 135^\circ$ is the expansion period. During this period the compression volume of the rhombic drive engine remains almost constant about zero. The piston and displacer move together downwards and the most of the working gas expands in the hot volume above the displacer. This is an advantageous property of the rhombic-drive engine to increase work generation. In lever drive engine, during the expansion period, the compression volume displays a bit negative and a bit positive variation.

The period $135^\circ \leq \theta \leq 225^\circ$ is cooling period at constant volume. In the lever-drive engine during this period the total volume is almost constant. The working gas travels from the hot volume to the cold volume while the total volume is constant. The piston generates no positive or negative work and the process is compatible with standard Stirling cycle. As seen from Fig. 4b, during the period $135^\circ \leq \theta \leq 225^\circ$, in rhombic drive engine the total volume exhibits a significant amount of variation and the piston generates positive work.

The period $225^\circ \leq \theta \leq 315^\circ$ is the compression period. As seen from Fig. 4a, in lever drive engine while the compression volume decreases, the expansion volume remains at about zero. This is a desired occurrence for minimum compression work. As seen from Fig. 4b, in rhombic drive engine, at the beginning of the compression period, the expansion volume is relatively larger. This causes that a considerable amount of the working fluid is compressed in the expansion volume which may result in a relatively larger compression work. However, in the rhombic drive engine, during the compression period some of the working fluid displaces from the expansion volume to the compression volume. In practice, this motion may cause positive effect via generating larger heat transfer coefficient in compression volume.

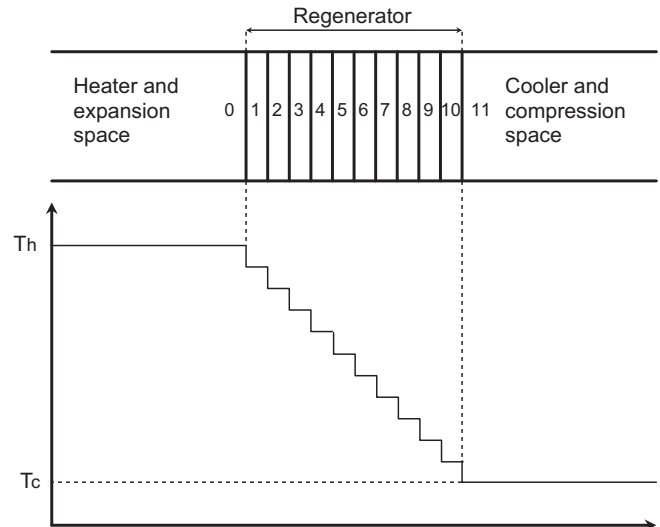
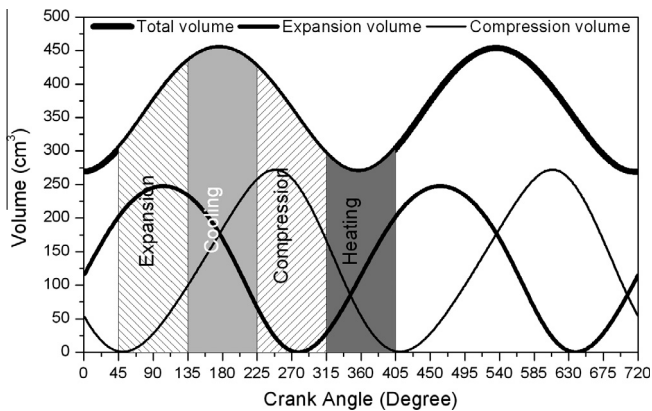
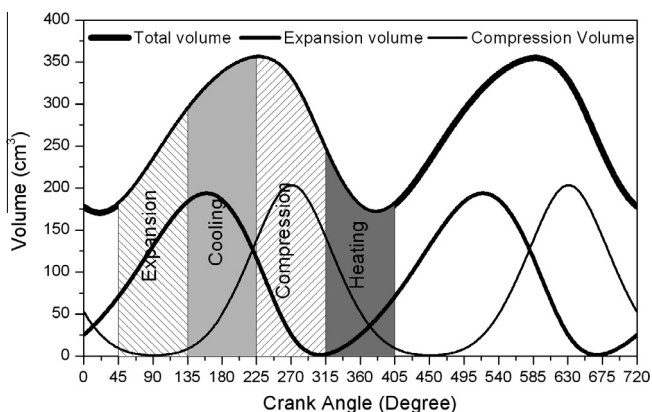


Fig. 3. Boundary temperature distribution.

Table 1

Specific values used in the case study.

	Lever	Rhombic
Gas constant of working fluid	2077 J/kg K	2077 J/kg K
Cold end temperature	350 K	350 K
Hot end temperature	850 K	850 K
Regenerator temperature	500 K	500 K
Piston stroke	6 cm	6 cm
Cylinder diameter	0.064 m	0.064 m
Piston rod length	0.1847 m	0.234 m
Displacer rod length	0.203 m	0.4153 m
Displacer rod area	2 cm ²	2 cm ²
Displacer length	0.1 m	0.15 m
Piston length	0.052 m	0.1 m
Lever x arm length	0.125706 m	–
Lever d arm length	0.098718 m	–
Intermediate arm length (h)	0.118 m	–
Lever arms angle	87.5°	–
Vertical distance between crank centre and lever centre (u)	0.07492 m	–
Horizontal distance between crank centre and lever centre (z)	0.119435 m	–
Side length of Rhombic	–	6.66 cm
Heater heat transfer area	200 cm ²	200 cm ²
Regenerator heat transfer area	1000 cm ²	1000 cm ²
Cooler heat transfer area	200 cm ²	200 cm ²
Engine speed	1200 rpm	1200 rpm
Time step used in analysis	0.00013 s	0.00013 s
Angular step used in analysis	$\pi/180$ rad	$\pi/180$ rad

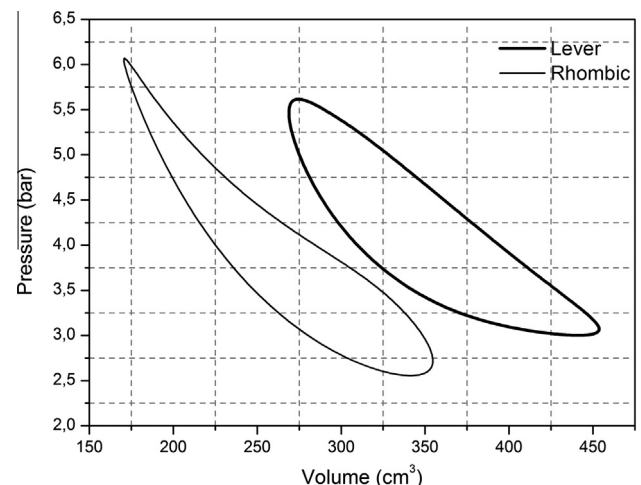
**Fig. 4a.** Kinematic behaviors of the lever drive engine.**Fig. 4b.** Kinematic behaviors of the rhombic drive engine.

The period $315^\circ \leq \theta \leq 405^\circ$ is the heating period. In lever drive engine the variations of the compression and expansion volumes are almost equal and the heating process occurs at constant volume. In rhombic drive engine however, during the heating process the compression continues. Decrease of the compression volume is larger than the increase of the expansion volume which causes negative work as well as restricting the heat transfer to the working fluid.

In Fig. 5 the P–V diagrams of lever drive and rhombic drive engines were compared. The data used for both diagrams are obtained for 500 W/m² K heat transfer coefficient and 4 bars charge pressure which is optimum value of the charge pressure for the condition used in this case study. Work provided by lever drive and rhombic drive engines are 17.02 and 14.38 W respectively. As seen from Fig. 5 the upper end of the P–V diagram given for rhombic drive engine is too thin. This is a result of inadequate heating at constant volume which was noted above. In lever drive engine the performance of both heating and cooling processes are relatively better. The compression ratio of rhombic drive is higher than the lever drive engine. At the same charge pressure, the lever drive engine contains more working fluid compared to the rhombic drive and as the result of this it generates more work.

Fig. 6 illustrates power–charge pressure profiles of two engines. Profiles were obtained for three different heat transfer coefficients as 500, 1000 and 1500 W/m² K. For these heat transfer coefficients, the optimum charge pressures appear as 4, 8 and 12 bars respectively. At higher values of charge pressure the upper limit of the temperature of thermodynamic cycle decreases. As a result of this, the thermal efficiency and power decrease. As seen from Fig. 6, the power of rhombic-drive is lower than the lever drive engine. The difference between the specific powers of engines, calculated based on maximum inner volume of the cycle, is not very significant. Table 2 indicates specific powers of engines obtained for 500, 1000 and 1500 W/m² K heat transfer coefficients.

Fig. 7 illustrates thermal efficiency–charge pressure profiles of two engines. In the calculation of thermal efficiency, the work was divided with the heat received from the heater. In nodal analysis the working gas also receives heat from the regenerator. In nodal analysis unless balancing the heat exchange between the working gas and regenerator, the correct value of the heat received from the heater cannot be calculated. To balance the heat exchange of the regenerator, some minimal changes may be imposed to the heat transfer area of the heater and cooler. This change may also be made in the area of nodes taking place at the cold and hot end of the regenerator. At optimum charge pressures corresponding to

**Fig. 5.** Comparison of the P–V diagrams of lever driven and rhombic drive engine in same charge pressure (4 bar).

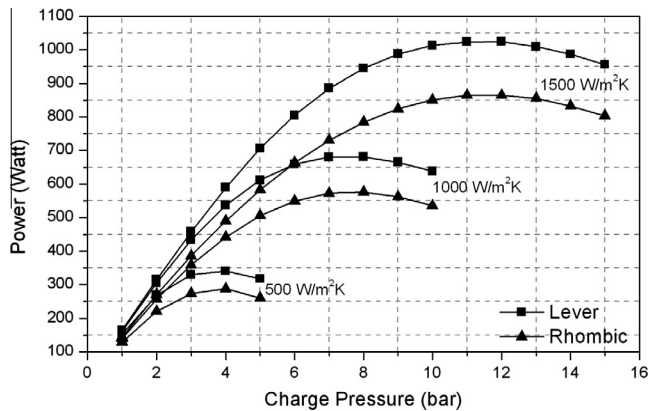


Fig. 6. Comparison of works generated by lever driven and rhombic drive engines.

Table 2
Specific powers of engines.

h (W/m ² K)	500	1000	1500
Lever (W/lt)	751.56	1501.27	2260.31
Rhombic (W/lt)	811.39	1624.7	2438.07

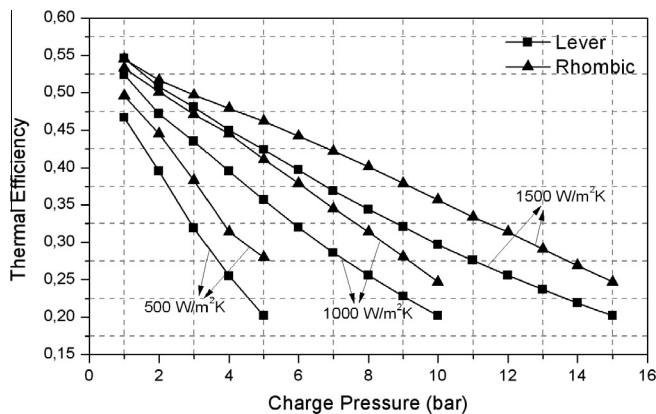


Fig. 7. Variations of thermal efficiencies with charge pressure and heat transfer coefficient.

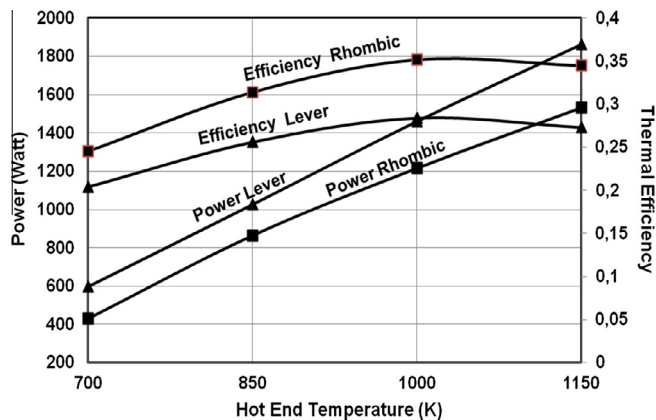


Fig. 8. Variation of thermal efficiency and power with hot end temperature.

500, 1000 and 1500 W/m² K heat transfer coefficients, the thermal efficiency of the lever drive engine appears to be 26%. For the same charge pressures the rhombic drive engine presents 31% thermal

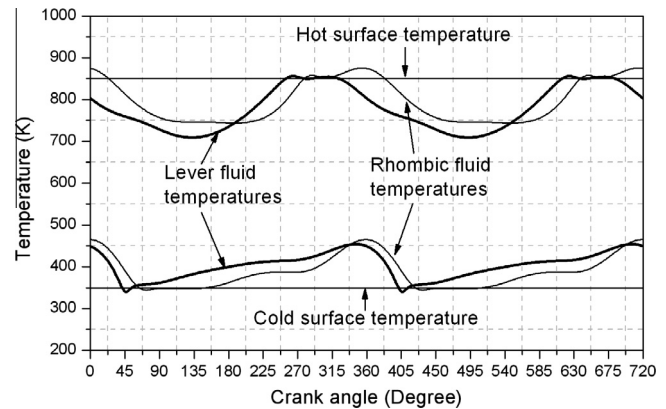


Fig. 9. Variation of fluid temperature with crank angle.

efficiency. At high values of heat transfer coefficient inherently the thermal efficiency is high. At high values of the charge pressure the thermal efficiency is low.

Fig. 8 illustrates variation of the thermal efficiency and power with the hot end temperature. Data used in Fig. 8 was obtained at optimum charge pressures corresponding to heat transfer coefficient of 1500 W/m² K. As seen from Fig. 8, the thermal efficiency has an optimum value at 1000 K hot end temperature. The decrease of the efficiency after 1000 K is caused by the inadequate heat transfer area and larger mass of the working fluid which restrict the upper and lower temperature limits of the thermodynamic cycle. The variation of power with the hot end temperature is more likely linear. Maximum thermal efficiency of the lever driven engine was obtained at 1000 K as 28% which corresponds to 1460 W power. Thermal efficiency of the rhombic drive engine at 1000 K was determined as 35% which corresponds to 1216 W power. For 350 K cold end temperature and 850 K hot end temperature the Carnot efficiency is 58%. The deficit between Carnot efficiency and above efficiencies is caused by the temperature difference between working gas in nodal volumes and the boundaries of them.

In Fig. 9, gas temperatures in expansion and compression volumes were compared with their boundary temperature for both engines. The inner heat transfer coefficient was taken as 1500 W/m² K. The charge pressure was 12 bar which corresponds to maximum power under 1500 W/m² K heat transfer coefficient. In the rhombic drive engine, the mass of working fluid is relatively less and, the difference between the gas temperature and the boundary temperature is lower than that of the lever driven engine. As the result of this, the thermal efficiency of the rhombic drive is a bit higher than that of the lever driven engine. Once the working fluid masses in both engines are made equal, more close thermal efficiencies seen in Fig. 10 are obtained. However, after a certain amount of working fluid mass, the efficiency of rhombic drive decreases below that of lever driven engine. Similarly, after a certain amount of working fluid mass the power of rhombic drive engine exhibits a quick decline. As the result, both engines have advantageous and disadvantageous working conditions.

5. Conclusion

A β type lever driven Stirling engine mechanism has been devised and its thermodynamic performance was compared with the rhombic drive engine's performance. The work generation of the lever driven engine was found to be higher than the rhombic drive engine on the same working conditions. Under the equal charge pressure the thermal efficiency of the lever driven engine

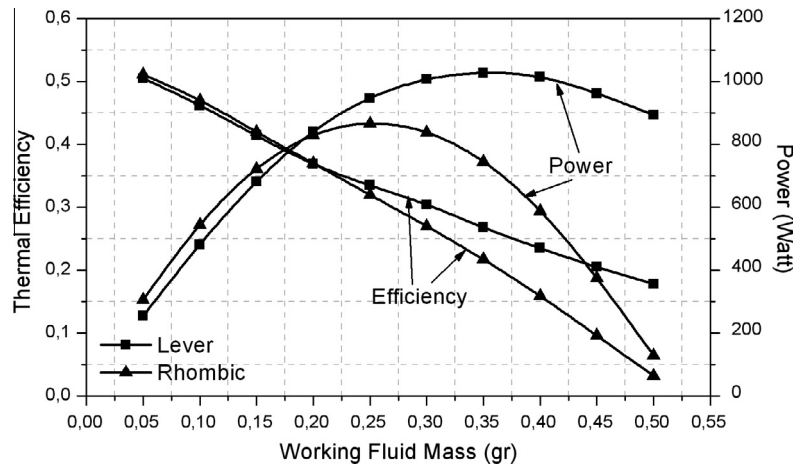


Fig. 10. Variation of power and thermal efficiency with working fluid mass ($h = 1500 \text{ W/m}^2 \text{ K}$).

was found to be lower than that of the rhombic drive engine. Under the equal mass of working fluid, the efficiency of lever driven engine is equal or higher than that of rhombic drive engine. The lever drive engine involves a lesser number of components compared to the rhombic drive engine as well as lower external volume and mass.

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